Observations on Dynamic Qualification Testing of a Component with Nonlinear Deadband Interfaces

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ABSTRACT

Typical flight hardware dynamic qualification tests exhibit nearly linear structural response when exposed to acceleration inputs from low to full qualification levels. Even in these "linear" cases, there is typically a trend of increasing damping with test levels. The nonlinear mechanism behind this increase in energy dissipation well understood – typically stick/slip hysteresis at joint connections.

This paper is concerned with a different type of nonlinearity: deadbands at structural interfaces. Deadband nonlinearities can have a significant influence on structural response and modal/spectral characteristics which can present difficulties in test, analysis, and structural certification. This subject nonlinear behaviour is observed during flight qualification testing of the AQUARIUS instrument and discussed here. Simple physical reasoning and analytical model is utilized to explain the behaviour which is consistent with the test findings.

1. INTRODUCTION

Recently AQUARIUS instrument, one of the NASA's missions managed by Jet Prolusion Laboratory's (JPL), underwent random vibration and acoustic qualification tests (Figure 1). The AQUARIUS mission was the first global observations of seas surface salinity, giving climatologists a better understanding of the ocean's role in Earth's water cycle and weather patterns. The instrument was designed to interface with the spacecraft using a series of bipods with mono ball joints and clevises on the instrument side. The random vibration tests conducted at JPL included implementation of instrument interface force limits to account for differences between the test versus the flight configurations impedances. The standard low-level broadband white-noise random surveys (0.45 grms from 10 Hz to 2000 Hz) was used before and after the instrument full qualification vibration level test. As the input to the instrument at the bipod interfaces was increased using requirements provided in Table 1, excessive chatters were observed on interfaces force and acceleration responses measured near the instrument bipods. The real-time test data analyses showed strong structural nonlinearity observed due to mono balls clearances and deadbands.

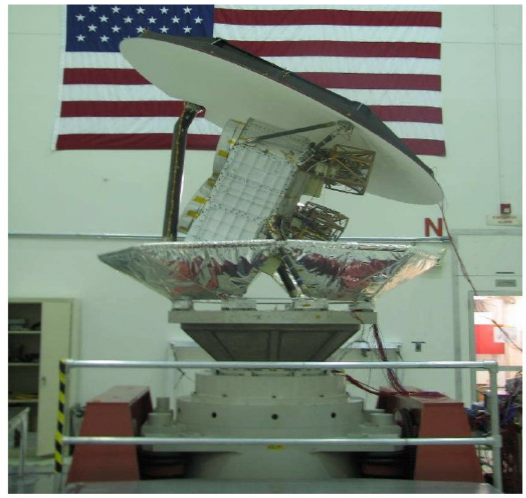


Figure 1 AQUARIUS Instrument mounted on to the shaker in vertical direction (Y-axis) that underwent random vibration tests. The interface and force gage locations are highlighted.

Axis	Frequency, Hz	Protoflight Level
X, Y	10	0.0125 g^2 / Hz
	10 – 20	+ 6 dB / Oct.
	20 – 200	0.05 g^2 / Hz
	200 – 400	- 6 dB / Oct.
	400	0.0125 g^2 / Hz
	Overall	3.78 g_{rms}
Z	10	0.00156g^2 / Hz
	10 – 20	+ 6 dB / Oct.
	20 – 200	0.00625 g^2 / Hz
	200 – 400	- 6 dB / Oct.
	400	0.00156 g^2 / Hz
	Overall	1.34 g_{rms}

Table 1 Instrument input acceleration in all three orthogonal axes.

Figure 2 depicts an acceleration time history of an accelerometer measured near one of these bipods. The last 60 seconds of this plot are from flight acceptance level random vibration responses in one of the instrument's lateral axes. As shown in this figure a peak loading as high as 50 sigma (peak/rms) was observed with significant peaks above 5-sigma over 60 seconds test duration. A typical linear structure would experience a couple of 5-sigmas over 60-second test period (Ref 1 and 2). Part of the higher than expected sigmas is attributed to deadbands and gapping of the ball joins and clevises. However, the degree in which peak loadings had occurred led us to believe that there are structural workmanship issues that needed to be addressed. The unusual chatter and shift in modal frequencies provided a challenging test to force limit the responses. The existence of extreme peaks also provided difficulty in identifying the instrument primary modes to satisfy flight frequency and loads requirements.

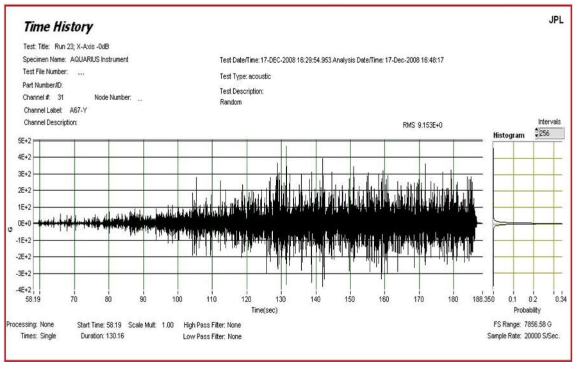


Figure 2 Acceleration time history measured near one of the mono-balls. The acceleration rms for full level random vibration test is estimated to be 8.9 where many extreme peaks above 5 sigma had occurred due to the deadband chatters (peak is 450+ g's)

The extremely nonlinear structural behaviour attributed to bipod interfaces (mono balls and clevises) – Figure 3. After examination of the joints it was discovered that mono balls had faulty gap tolerances that led to unusual structural nonlinear response behaviour. Issues related to the quality of the as-installed mono balls, chipping of the liner edges, installation and ball-to-liner tolerance, and potential for mono-ball-to-clevis gapping were discovered. The unexpected shift in modes are believed to be the result of the nonlinearity of the system, and were attributed to the main bipod joints and mono balls. Physical evidence of the interfaces also suggested that some of the joints were looser than others, which points to the flaws in workmanship.

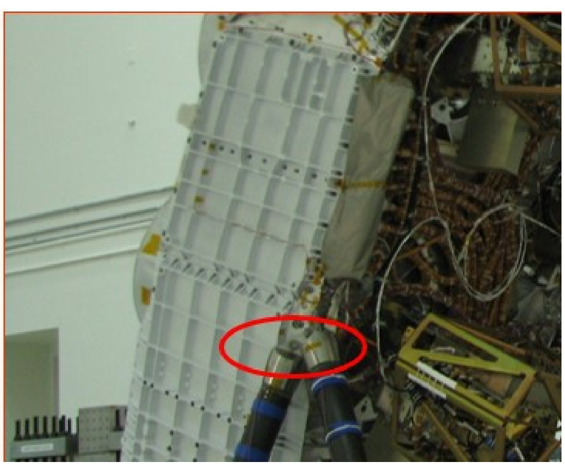


Figure 3 Significant gapping that resulted in chattering observed at all 12 Instrument bipod mono-ball mechanisms during random vibration testing.

Even though the random vibration test objectives were met and that both workmanship and the ability of the instrument and its components to withstand the launch environment with margin were demonstrated corrective actions were taken to a) eliminate excessive gaps on the joints of the main bipod including replacement of the lined mono balls with higher precision metal-to-metal mono balls, b) conduct a before and after joint modification hammer test, and c) conduct a post-modification vibration test to validate the effectiveness of changes made in ensuring instrument workmanship screening is properly met.

After the mono ball and clevis re-work and instrument random vibration test, the inherent gap in the ball and clevis joints provided the classical and predictable nonlinear structural dynamics behaviour (See Section 4).

2. PRETEST ANALYSIS

Typical pretest analysis involves the construction of a linear finite element model (FEM) and the execution of modal analyses (See Figure 4). Although this structure is highly nonlinear due to the presence of interface deadbands, linear modal analyses with (1) all interfaces constrained and (2) all interfaces free may shed some light into the bounding modal states relative to test levels. Still, a rigorous pretest analysis that is of high value to the testing must involve the modelling of the deadband nonlinearities and time-domain nonlinear simulations [Refs. 3, 4].

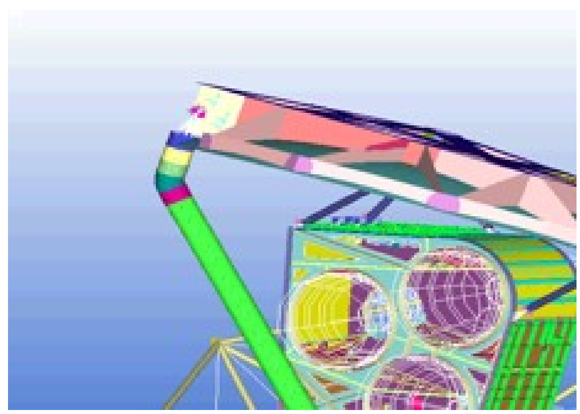


Figure 4 – Instrument Finite Element Model.

3. TEST RESULTS

The observations made on the instrument nonlinear behaviour, after mono ball modifications, are discussed in this section. The instrument random vibration was performed by mounting it to blocks atop force gauges in the upright shaker head expander (Y-axis) and the horizontal slip table (X- and Z-axis). For this test thirty-two accelerometers were attached to the test article to monitor critical responses on the instrument. Twenty-two force transducers were installed in between the vibration test fixture and the instrument (one for each mounting bolt), oriented parallel to the hardware coordinate axes, and torqued to appropriate values. The responses from force transducer were summed to obtain the total forces at the instrument interfaces. The instrument had gone through acoustic test, where details of the acoustic test is not provided in this paper but as a reference Figure 5 is the acceleration power spectral density (PSD) measured near one of the bipods. The deadband induced nonlinearity is not as prevalent in acoustic induced vibration as the acoustic energy is low below 100 Hz and it is not effective in displacement of the instrument at its interfaces. The instrument was subjected to the random vibration environment for 60 seconds to the environments indicated in Table 1. The total interface forces measured at the instrument interfaces were limited for each axis of testing to remove shaker configuration impedance related conservatisms.

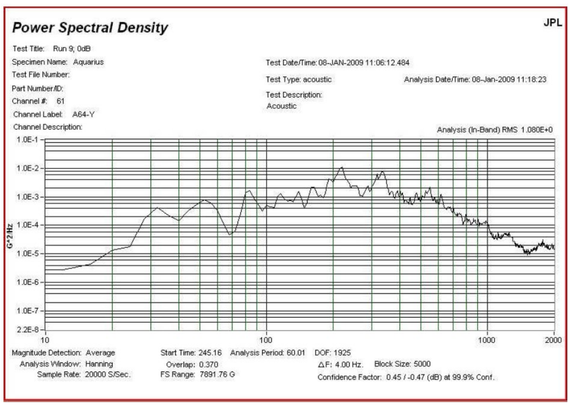


Figure 5 – Acceleration PSD measured near one of the bipods obtained from acoustic test.

Total interface force response spectral densities are plotted in Figure 6a-c for Z-, X-, and Y-axis, respectively. Each Figure shows the spectral shapes computed from low input levels (white-noise with 0.45 grms) to higher inputs with a 3 dB increment starting from 18 dB below the requirements shown in Table 1. The white-noise pre- and post-full level random vibration power spectral densities for the Z-axis are shown in Figure 7. The following observations are made from these power spectral densities. First, the pre- and post-full level PSD overlays for Z-axis indicates that the primary structural mode of ~40 Hz did not change after the hardware underwent full level random vibration excitation (Figure 7). However, as the input to the hardware increased the force spectral shape significantly changed with the appearance of two bi-modal excitation near 16 Hz, and 26 Hz. These are the product of the nonlinear system behaviour due to gapping at the mono ball interfaces. The ~40 Hz mode is shifted downward and is damped significantly. Second, further increase in input levels did not cause further change in spectral characteristics as are demonstrated in Figure 6a for this axis. Third, Figures 6b and 6c for other two axes (axial Y-axis and lateral X-axis) indicate similar structural behaviour. These are related to the classical nonlinear deadband phenomenon (Section 4).

Further data analysis is performed to gain more insight into the nonlinear structural dynamics. The interface force FFTs over a few seconds selected from different times over the random vibration test period were estimated and are plotted in Figure 8. As shown in Figure 6a and 6b the instrument shift in frequency due to the gap is stable. The FFTs for these plots are taken from -9 dB Z-axis test data. Figure 8a is FFTs of the first several seconds and Figure 8b are taken from last several seconds of the test data.

A series of time histories of the interface forces in one of the lateral directions (Z-axis) are shown in Figure 9. The first plot from the top (Figure 9a) is taken from a white-noise low level random vibration test with forces summed over all interfaces. The fundamental mode for this low input level is approximately 40 Hz with PSD shown in Figure 7. The random vibration responses shown in Figure 9a appears to be semi-Gaussian. The departure from the normal distribution of the random responses indicates the impact of the gap is already being felt at the mono ball interfaces. The next two plots (Figure 9b and c) are taken from -18 dB of the full level (See Table 1) for the same axis of excitation. Figure 9b is forces summed over all bipod interface responses, and Figure 9c is the force response from one of the bipods interface force only. Again the PSD for this case is shown in Figure 6, where the impact of the deadband on structural behavior is significant and the shift in frequency had already occurred. The asymmetric non-Gaussian time history from one of the bipods shown in Figure 9c clearly indicates that the structural excitation are in deadband zone. The time-history for -15 dB and -9 dB also shown in Figure 9d and e, respectively. The increase in number of chatter and in extreme peaks for these plots qualitatively indicate the displacements of the structures within the mono ball gaps are occurring more frequently (i.e. with faster speed). The PSD densities above -18 dB as are shown in Figure 6 indicates the structural stiffness change cease to exist (no change in modal damping and frequencies). The transition of the slow structural movement to fast movement within the gap from the low level input to higher inputs using a simple analytical model are discussed in next section.

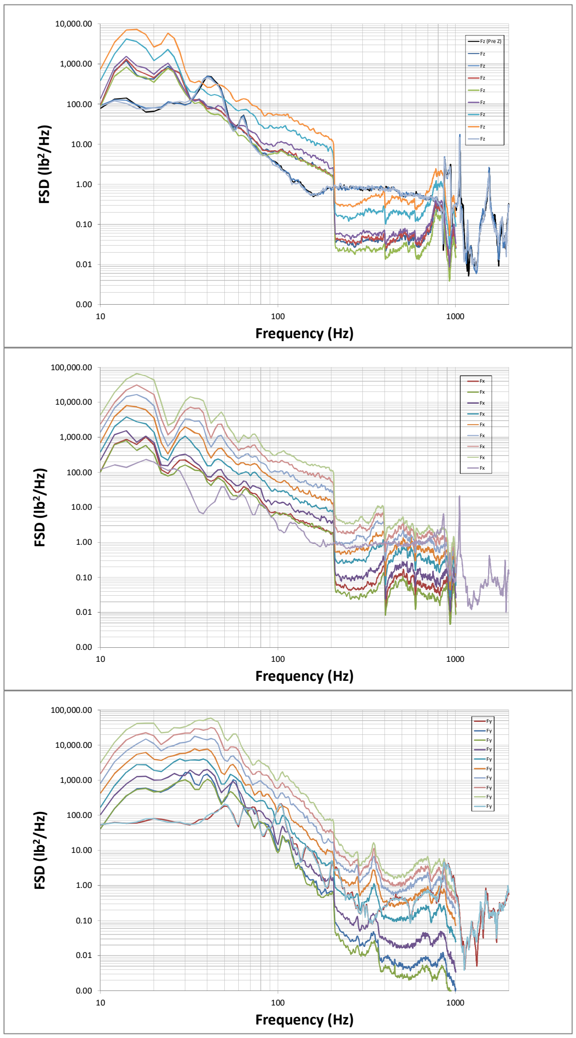


Figure 6a-c – Summed interface force power spectral densities from low input level to higher levels shown for three orthogonal axes.

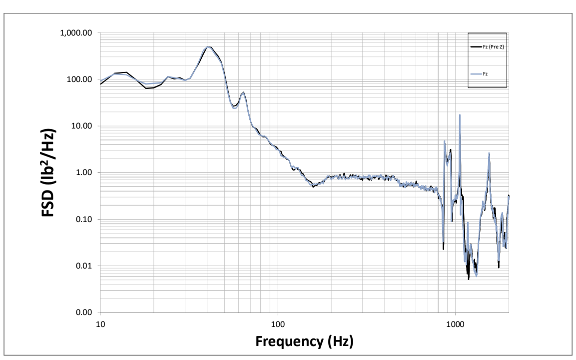


Figure 7 – Summed interface force power spectral densities comparison of pre- and post-full level in the Z direction.

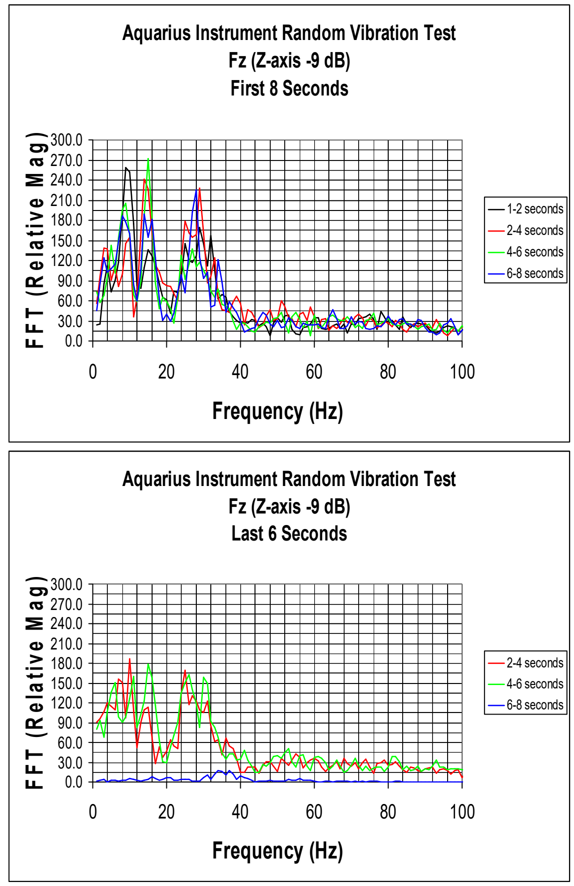
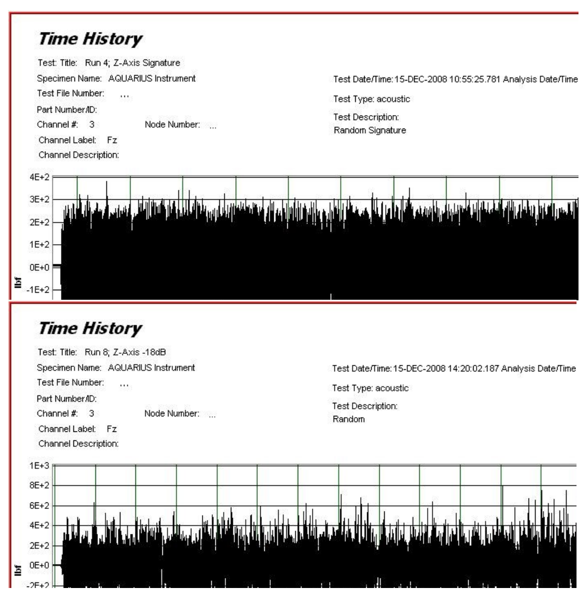


Figure 8a-b – Force FFTs estimated over 2-second period for different times a) at the beginning of the run and b) towards the end of the random vibration test (-9 dB Z-axis).



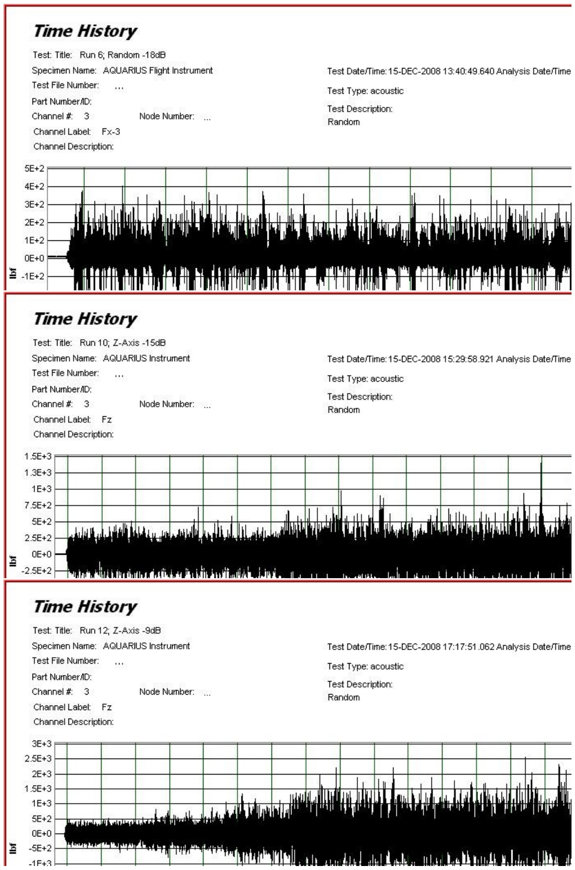


Figure 9a-e – Force response time histories measured at the instrument interfaces in the Z-direction.

4. EXPECTED DEADBAND BEHAVIOUR VS TEST LEVEL

Consider a component supported at multiple interfaces which include deadbands. Assume each deadband has a displacement limit [-d, +d]. Each deadband possess 3-states: bottomed out at the -d and reacting a positive force, bottomed out at the +d and reacting a negative force, and in transitioning between the two limits and reacting zero force (assuming a pure deadband with no stick/slip friction). To demonstrate the complexity of such a nonlinear system, assume the component is supported at 4 interfaces. Then the component will possess 3^4 or 81 possible modal states – a complex nonlinear system. However, some simple reasoning, backed by both nonlinear simulations and test, can be used to explain the behaviour of systems inclusive of deadbands relative to test levels.

In a low level test, with "low" defined relative to the deadband limits, the interfaces are transitioning relatively slower between the two limits, therefore, the amount of time spent at zero interface forces becomes longer. With this, the component behaves as if the boundary conditions were free (non-force reacting). At higher test levels, again with "higher" defined relative to the deadband limits, the interfaces will transition faster and therefore the amount of time spent in transition (i.e., zero force state) becomes shorter. In this scenario, the component behaves more "linear" with force reacting boundary conditions. In addition, it follows from the same reasoning that any further increase in test levels would not modify this linear behaviour of the deadband nonlinearities.

To quantify the effect of test level on natural frequency, consider a cantilever beam supported at a deadband interface. Utilizing above reasoning, at the lower test levels, the cantilever's fundamental bending mode will resemble the bending mode of a free-free beam. At higher test levels, the same mode will more closely adhere to the fundamental cantilevered bending mode.

The fourth order partial differential equation (PDE) for an Euler beam is given as follows:

$$EI(x)\frac{\partial^4 w(x,t)}{\partial x^4}=\mu($$

where w(x,t) represents the lateral displacements as a function of space and time, EI represents the bending stiffness parameter, and μ the mass per unit length. We know from the eigenvalue problem solution of the above PDE that the fundamental bending frequency of a free-free beam is roughly a factor of 6 higher than the same beam cantilevered. Therefore, there is a drop in frequency associated with increase in test levels up to a fully linear behaviour at which the frequency would plateau.

The above physical reasoning can be summarized as follows: for a component supported by a set of interfaces inclusive of deadband nonlinearities, as is the case for the AQUARIUS instrument, the expectation would be for a drop in primary modal natural frequency with increased test levels with the frequency/spectral characteristics stabilizing at the higher test levels. The instrument test is more complex than the simple illustrative example of the cantilevered beam so test and/or high fidelity nonlinear simulations would be required to quantify the modal/spectral characteristics as a function of test level.

5. SUMMARY

As seen in the AQUARIUS instrument dynamic qualification tests, deadbands can have a significant influence on increasing structural response and changing modal/spectral characteristics. In the instrument test, the fundamental frequency of the test article dropped from 40 to 16 Hz with increasing test levels. Once the test level was "high enough" (relative to deadband limits), the fundamental frequency "stabilized" at 16 Hz with no further changes in modal/spectral characteristics. This is consistent with the expected deadband behaviour as described in Section 4 and nonlinear simulation findings in Ref. 3. It is recommended that the pretest analysis for components involving deadband interfaces include time-domain nonlinear simulations.

6. REFERENCES

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7. AKNOLEDGMENT

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